



Role of surface radiation on the functionality of thermoelectric cooler with heat sink



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HIGHLIGHTS

- Nowhere the performance of TEC has been evaluated considering the heat sink to be emitting thermal radiation.
- There is a shift in the operating points of the TEC for all level of currents when radiation is considered.
- The non-functional zone comes closer to the operating zone for radiation for the second working temperature range.

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ABSTRACT

The present investigation substantiates the inclusion of radiation heat transfer coming from heat sink while modelling the thermal performance of a thermo-electric cooler (TEC). A commercial grade thermoelectric cooler from Laird Technology and a heat sink from Aavid Thermal Alloy are chosen to perform this study and their actual values revealed by the respective company datasheets have been considered for the comparison. The results provided in the company datasheets are based on the assumption that the heat sink is mounted on the TEC and is cooled by natural convection only. Therefore, the present study conducted in this work, also identifies the heat transfer solely due to natural convection. The results are validated against the available operational company datasheets. Subsequently, the effect of surface radiation from the heat sink in addition to the natural convection on the thermal performance of TEC has been investigated. Furthermore, to bring more clarity in the investigation, two different ranges of working temperatures have been opted and shift in the operational points because of radiation in both ranges have been delineated through data tables, figures and comparative plots.

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1. Introduction

In spite of high cost and poor power efficiency, thermoelectric cooling is appreciated as it is considered to be an environment friendly refrigeration technology i.e. “Green refrigeration technology” for small scale localized cooling applications such as in computers, infrared detectors, electronics and optoelectronics applications. A thermoelectric refrigerator is a solid-state active cooler which transfers heat from the cold junction to the hot junction connected by two different semiconductor materials continuously being actuated by the direction of the current (widely known as Peltier effect).

Relevant few literatures have been referred in this context and are discussed in the following. Yang et al. [1] have described the transient response of TEC with and without mass load through examination of both the minimum temperature reached and the time constant involved in the cooling and recovering stages. Chang et al. [2] investigated that the total thermal resistance of the thermoelectric cooler (R_{TEC}) increases with heat load and decreases with input current, whereas thermal resistance in the heat sink (R_{hs}) increases with input current and decreases with heat load. In their study, they further evaluated the overall resistance (i.e. $R_t = R_{TEC} + R_{hs}$) under every heating power and input current. Due to contrary trends of R_{TEC} and R_{hs} for input current, they could find an existence of optimal input current for each R_t . Huang and Duang [3] theoretically have solved a linear dynamic model of the thermoelectric cooler including the heat sink and the cooling load heat exchanger using small signal linearization method. Yushanov et al [4] have numerically solved the governing equations related to the

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Nomenclature		W	Width of heat sink base (m)
A	Exposed surface area to ambient (m ²)	Z	Figure of merit (K ⁻¹)
C	k*G (W/K)	<i>Greek alphabets</i>	
COP	Coefficient of performance	α	Seebeck coefficient (V/K)
F	View factor	ρ	Electrical resistivity (Ω m)
G	Geometry factor (m)	η	Overall surface efficiency of heat sink fins
h	Heat transfer coefficient (W/m ² K)	ε	Emissivity of the heat sink surface
H	Height of the heat sink fin (m)	σ	Stefan–Boltzmann constant (5.67×10^{-8}) (W/m ² K ⁴)
I	Applied current at TEC (A)	<i>Subscripts</i>	
k	Thermal conductivity of air (W/mK)	a	Ambient
L	Length of the heat sink fin (m)	b	Heat sink base
N	Number of n-type & p-type couples in the TEC	c	convective
Nu	Nusselt number	ch	Heat sink channel
n	Number of fins of the heat sink	h	Hot side of TEC
Q	Heat transferred (W)	hs	Heat sink
R	Thermal resistance (K/W)	l	cold side of TEC
R'	Electrical resistance of the TE material (Ω)	nc	Numerical code
Ra	Rayleigh number	r	radiative
S	Gap between fins of the heat sink (m)	sa	Sink to ambient
T	Absolute temperature (K)	t	Total
t	Thickness (m)		

thermoelectric phenomenon using COMSOL multi physics commercial tool, where they studied the effect of the temperature dependent material properties of the p and n type semiconductors. Felgner and Frey [5] have shown a transient thermal analysis of both thermoelectric coolers and thermoelectric generator built in Modelica language where it has been considered that the material properties vary with 1D spatial temperature distribution. Lossec et al [6] have demonstrated how the electricity produced from human body by attaching a thermoelectric generator can be maximized and with regard to this, they introduced a new factor Z_E which takes into account not only the physical characteristics of the thermoelectric materials but also the quality of thermal coupling involved.

But, in none of the papers cited above, the performance of TEC has been evaluated considering the heat sink fins to be emitting thermal radiation.

Therefore, this study is focussed to trace the crucial effects that a TEC encounters due to thermal radiation by the attached heat sink.

2. Governing equations and computational procedure

A thermoelectric cooler Fig. 1a consists of a thermoelectric module, a heat sink connected to the hot side, where as a heat exchanger connected to the cold side subjected to a cooling load. The thermoelectric module comprises of many pairs of p-n type thermoelectric material connected in series and clamped and soldered with two base plates. The cooling load Q_l is absorbed at the cooling end of the heat exchanger and subsequently conducted to the hot end plate (i.e. hot side of the TEC).

The following Equations (1) and (2) express heat load at the two junctions, which are given as:

$$Q_l = N \left\{ \alpha T_l I - \frac{1}{2} I^2 R' - C(T_h - T_l) \right\} \quad (1)$$

$$Q_h = N \left\{ \alpha T_h I + \frac{1}{2} I^2 R' - C(T_h - T_l) \right\} \quad (2)$$

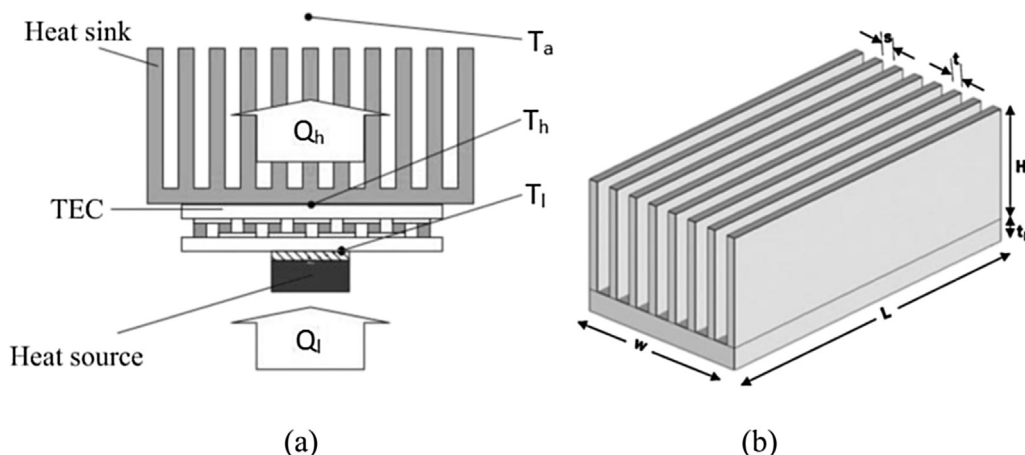


Fig. 1. Basic schematic of a thermoelectric cooler with heat sink.

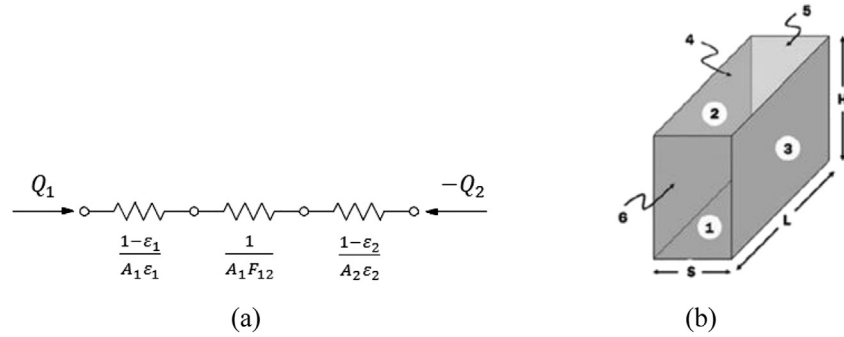


Fig. 2. (a) Radiative resistances for an infinite parallel plate, (b) Channel of a heat sink with finite length.

The surfaces of the heat sink shown in Fig. 1b are assumed to be at constant temperature because of high thermal conductivity and the heat is primarily carried by the air medium. With this configuration, earlier workers mostly have considered natural convection as the sole mode of heat transfer. It is well known that for the associated natural convection phenomenon, both the non-linear momentum equations (N–S equations) and energy equations get coupled through their source terms. The continuity equation along with energy equations and momentum equations when get solved, helps to quantify the heat transfer. Obviously, for this type of natural convection problem, the Nusselt number happens to be the function of the Rayleigh number. In this regard, Incropera and Dewitt [7] have established an empirical relation for quantifying heat transfer through natural convection as a function of geometrical factors and Rayleigh number through following Equations (3) and (4).

$$Nu = \frac{hS}{k} = \left[\frac{576}{\{Ra(\frac{S}{L})\}^2} + \frac{2.873}{\{Ra(\frac{S}{L})\}^{0.5}} \right]^{-0.5} \quad (3)$$

$$Q_{hs,c} = h(2nLH)(T_h - T_a) \quad (4)$$

However, in reality, thermal radiation from the heat sink surfaces does exist and needs to be accounted. The concept of thermal resistance (i.e. combined effect of surface resistance and space resistance) offered to the radiative heat transfer between two parallel infinite plates with reference to M.F. Modest [8] can be well conceptualized with the help of Fig. 2a.

The amount of radiation heat transfer exchanged between the two surfaces maintained at T_1 and T_2 are written as

$$Q_1 = \frac{\sigma(T_1^4 - T_2^4)}{\frac{1-\epsilon_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{1-\epsilon_2}{A_2 \epsilon_2}} = -Q_2 \quad (5)$$

Table 1
Comparison of output data of the developed code against AzTEC software.

R_{sa}	T_h	Q_c	Q_h	COP	Q_{h-nc}	Q_{c-nc}	COP_{nc}	% diff
0.08	293.83	20.12	35.35	1.32	35.64	20.16	1.30	1.34
0.10	294.55	19.93	35.47	1.28	35.51	19.96	1.28	0.24
0.20	298.00	18.96	34.99	1.18	34.89	18.98	1.19	1.16
0.30	301.36	18.04	34.54	1.09	34.28	18.03	1.11	1.84
0.40	304.64	17.13	34.09	1.01	33.68	17.10	1.03	2.15
0.50	307.83	16.25	33.67	0.93	33.10	16.20	0.96	3.05
0.60	310.95	15.39	33.25	0.86	32.53	15.31	0.89	3.39
0.70	314.00	14.57	32.85	0.80	31.97	14.44	0.82	2.97
0.80	316.97	13.75	32.47	0.73	31.42	13.59	0.76	4.42
0.90	319.88	12.95	32.09	0.68	30.88	12.76	0.70	3.52

In Fig. 2b, a typical channel is shown formed by two adjacent fins of a plate-fin heat sink. The Equation (5) needs modification for a finite geometry of the parallel plates, which has been discussed in the article of Younes Shabany [9].

Considering the channel surface of finite size to be diffuse and gray, and the surrounding medium comparatively large enough, the net radiation heat transfer rate from this channel can be expressed as

$$Q_{ch} = \frac{\sigma(S + 2H)L(T_h^4 - T_a^4)}{\frac{1-\epsilon}{\epsilon} + \frac{1}{F_{sa}}} \quad (6)$$

where F_{sa} represents the channel view factor which is the total view factor between the walls and the base of the channel and its surrounding.

Assuming the heat sink temperature to be T_h , the total radiation heat transfer rate from the heat sink is expressed as

$$Q_{total} = (n-1)Q_{ch} + [nt(L+2H) + 2HL + 2t_b(L+W)]\sigma\epsilon(T_h^4 - T_a^4) \quad (7)$$

Following the above expression, the equivalent radiation heat transfer coefficient (h_r) can be calculated as;

$$h_r = \frac{Q_{total}}{A_{hs}(T_h - T_a)} \quad (8)$$

A more accurate estimation for the actual radiation heat transfer rate can be made by considering the fin equations to account and is expressed as

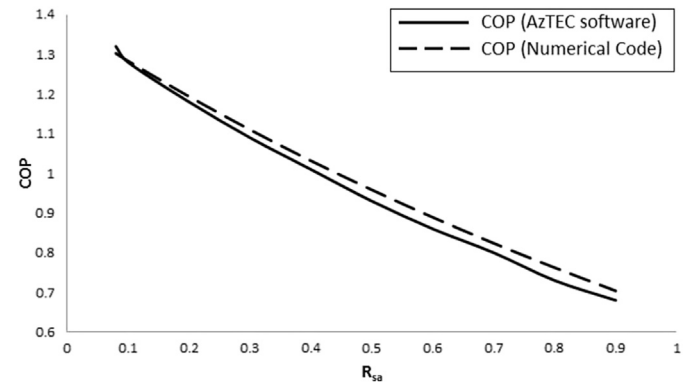


Fig. 3. Variation of COP with R_{sa} and its validation with the results obtained using AzTEC software.

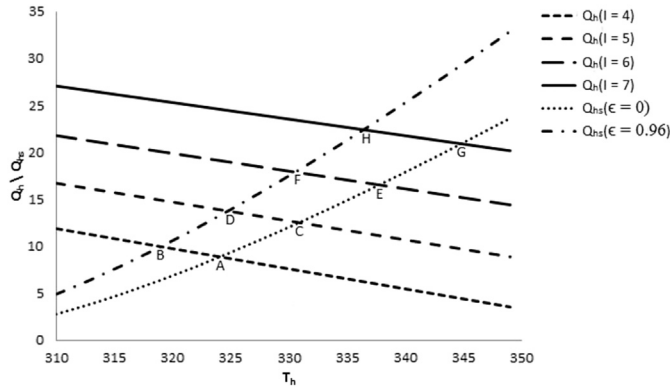


Fig. 4. Operating points with and without radiation when T_h varies from 310 K to 349 K.

$$Q_{hs,r} = \eta_{hs} A_{hs} h_r (T_h - T_a) \quad (9)$$

So, the total heat dissipated by the heat sink can be simplified as;

$$Q_{hs} = Q_{hs,c} + Q_{hs,r} \quad (10)$$

3. Numerical validation and results and discussions

Before discussing the effect of surface radiation on thermal performance of TEC, the validation of the results obtained from the present algorithm has been made. A numerical code has been developed in SCILAB to solve the above set of equations under the specified boundary conditions to capture the behaviour of TEC mounted with a plate fin heat sink. At first, to validate the numerical code, a standard commercial TEC (i.e. CP2,31,06L from Laird Technology database) is chosen and is subjected to a simple performance analysis without considering radiation. The input variables considered for the analysis are $T_a = 291$ K, $T_l = 288$ K, $I = 8.19$ A and R_{sa} is varied from 0.08 to 0.9 W/K to replicate various heat sink geometries to be mounted on the TEC. From the datasheet of CP2,31,06L, the parameters with assigned values (i.e. $G = 0.282$,

$C = 4.25 \times 10^{-3}$ W/K, $\rho = 1.01 \times 10^{-5}$ Ω m, $\alpha = 2 \times 10^{-4}$ V/K, $R' = \rho/G = 3.58 \times 10^{-3}$ Ω) are used in the calculations.

After running with the above inputs on both AzTEC software (i.e. a tool developed by Laird Technologies Ltd.) and the in-house code, the results obtained are shown in Table 1.

The COP variation with increasing R_{sa} obtained from in-house code is compared with results predicted from AzTEC software and is presented through Table 1 and Fig. 3. They are found to be in a good agreement with each other. However, the disagreement (given as % difference) found to be more towards the higher values of heat sink resistance. As the thermal resistance of the heat sink increases, the temperature gradients across its surfaces also increases, causing deviation in the assumption of isothermal condition made for the heat sink.

After the authentication of the present code, the effect of thermal radiation from heat sink exposed surfaces on the functioning of thermoelectric coolers has been discussed in the following section. For the purpose, the ambient temperature is kept fixed at $T_a = 295$ K and other parameters remain unchanged. Various commercial heat sinks from Aavid Thermal Alloy are selected and simulated along with the TEC using the present code to study the changes in COP. A number of output data are collected and studied extensively to understand the contribution of thermal radiation in TE refrigeration process. To present the important attributes of the study, a black anodized ($\epsilon = 0.96$) coated aluminium heat sink (No. 76620) is mounted on the selected TEC and the data are collected basically for two ranges of T_h values. The first range of T_h is chosen as 310 K–349 K maintaining T_l at 290 K, whereas the second range of T_h is chosen as 300 K–345 K maintaining T_l at 280 K.

In the first range, the results are obtained for four different current values i.e. 4 A, 5 A, 6 A and 7 A keeping the other parameters invariant. Below 4 A current, the TEC cold side is unable to withdraw heat from the object considering the present settings. For each current value, two operating points are obtained for the TEC corresponding to conditions when heat is lost from the heat sink through (i) natural convection only and (ii) combined natural convection and radiation. The operating points are plotted for all the four current values and are shown in Fig. 4.

The operating condition of any TEC i.e. for a particular value of current flow and a fixed temperature of cold end, heat lost from the heat sink and its temperature end get fixed, satisfying the principles of both thermodynamics and heat transfer. In Fig. 5,

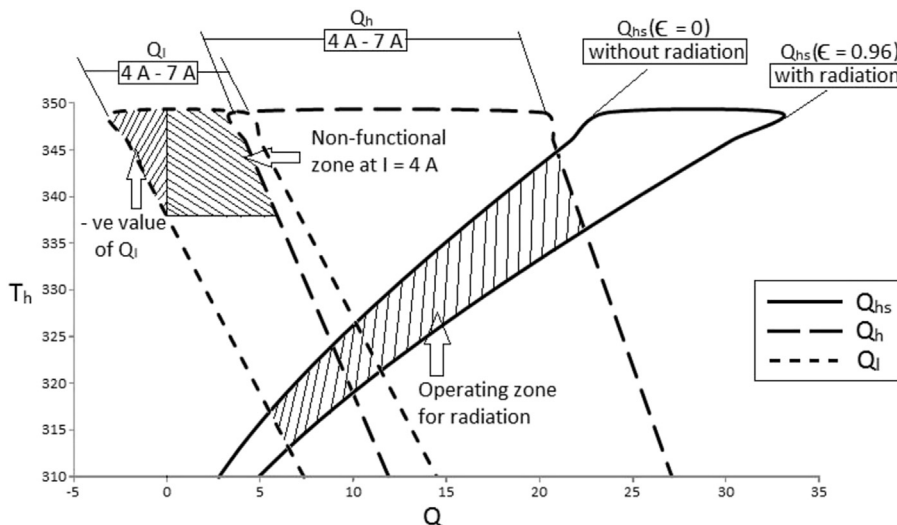
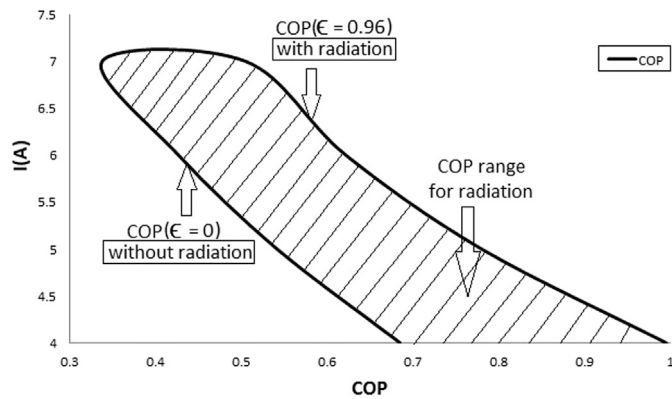


Fig. 5. Operating zone of TEC when T_h varies from 310 K to 349 K.

Table 2Variation in Q_h and Q_l with different values of current, when T_h lies in the range 310 K–349 K.

T_h	Thermoelectric cooler (CP2,31,06)								Heat sink (76620)		
	I = 4		I = 5		I = 6		I = 7		Q_{hs} (conv)	Q_{hs} (rad)	Q_{hs} (total)
	Q_h	Q_l	Q_h	Q_l	Q_h	Q_l	Q_h	Q_l			
310.00	11.88	7.34	16.72	9.94	21.79	12.31	27.08	14.46	2.78	2.12	4.90
313.00	11.24	6.55	16.12	9.15	21.22	11.52	26.55	13.67	3.86	2.59	6.45
316.00	10.60	5.76	15.52	8.35	20.65	10.73	26.02	12.88	5.08	3.06	8.14
319.00	9.96	4.97	14.91	7.56	20.09	9.94	25.49	12.09	6.41	3.55	9.96
322.00	9.31	4.18	14.31	6.77	19.52	9.15	24.96	11.30	7.83	4.06	11.89
325.00	8.67	3.39	13.70	5.98	18.95	8.36	24.43	10.51	9.34	4.58	13.92
328.00	8.03	2.60	13.10	5.19	18.39	7.57	23.90	9.72	10.93	5.11	16.04
331.00	7.39	1.80	12.49	4.40	17.82	6.78	23.37	8.93	12.59	5.66	18.25
334.00	6.75	1.01	11.89	3.61	17.25	5.99	22.84	8.14	14.31	6.22	20.53
337.00	6.11	0.22	11.28	2.82	16.68	5.20	22.31	7.35	16.08	6.80	22.88
340.00	5.46	−0.57	10.68	2.03	16.12	4.41	21.78	6.56	17.91	7.39	25.30
343.00	4.82	−1.36	10.08	1.24	15.55	3.62	21.24	5.77	19.78	8.00	27.78
346.00	4.18	−2.15	9.47	0.45	14.98	2.82	20.71	4.98	21.69	8.63	30.31
349.00	3.54	−2.94	8.87	−0.34	14.41	2.03	20.18	4.19	23.63	9.27	32.90

**Fig. 6.** Variation COP with current values when T_h lies in the range from 310 K to 349 K.

intersection points A, C, E and F represent the operating points of TEC corresponding to the flow of current of 4 A, 5 A, 6 A and 7 A respectively, when heat is lost through natural convection only. Similarly, B, D, F and H represent the operating points of TEC, when its heat sink lost heat through combined natural convection and radiation. For all current levels, it is observed that there is a shift in the operating points (i.e. from A to B, C to D, E to F and G to H) leading to drop in T_h value, when heat loss through radiation has been considered.

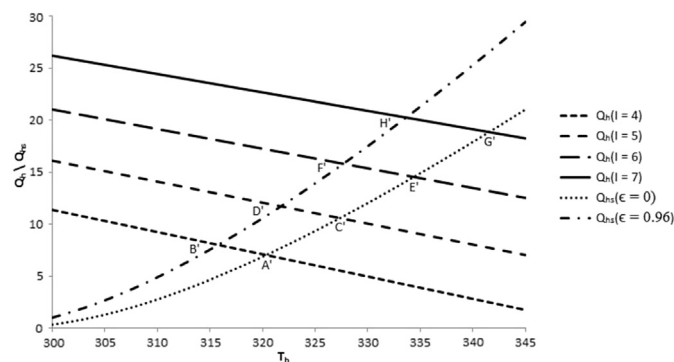
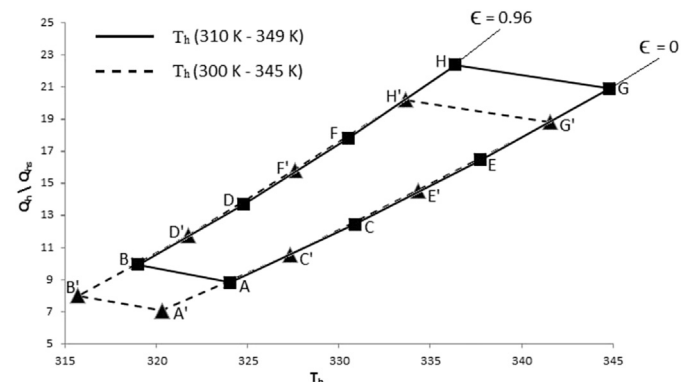
**Fig. 7.** Operating points with and without radiation when T_h varies from 300 K to 345 K.

Fig. 5 has been presented in order to identify the operating zone for TEC in absence of radiation and in presence of radiation.

In the above figure, it is observed that the operating zone for radiation with the existing conditions applied to the TEC and the heat sink, is lying in a safe range of positive heat transfer values. With the present setup, major non-functional zone of the TEC can be found at a current of 4 A when the hot side temperature exceeds 338 K, which means that the cold side of the TEC is rejecting heat to the object making the Q_l value negative beyond this temperature. With the increase in the value of current, the value of Q_l is marching towards a positive value and exactly at a current of 5.15 A, the value of Q_l steps into the positive range completely. It can be further seen that with higher value of T_h , the width of the radiation operating zone becomes wider along the same current lines, implying significant role of radiation in high temperature ranges. The above statement is further clarified from data provided in **Table 2**. The TEC performance is thereby evaluated from the above table and is shown in **Fig. 6**.

It is well understood from the above figure that the absolute value of COP becomes higher as the operating current decreases for both radiating as well as non-radiating cases whereas it decreases substantially as the current value rises to higher values. Moreover, the bandwidth of COP, for emissivity $\epsilon = 0$ to $\epsilon = 0.96$, seems to be more at lower current values which means that presence of radiation becomes more advantageous with respect to the performance of TEC at lower currents. The bandwidth becomes narrower and COP becomes less with the increase in the value of current which

**Fig. 8.** Recorded drift in the operating points for the two T_h ranges studied.

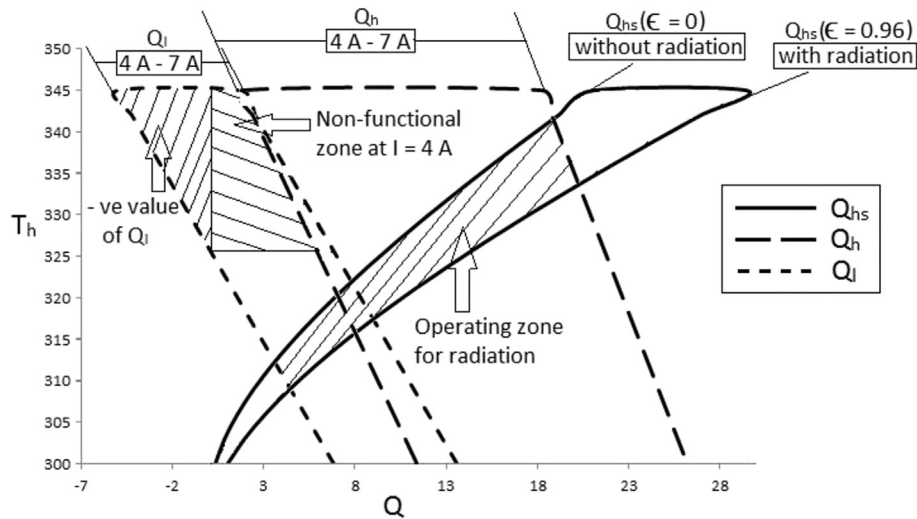


Fig. 9. Position of radiation operating zone inside Q_h/T_h plot when T_h varies from 300 K to 345 K.

Table 3

Variation in Q_h and Q_l values at different values of current for T_h range 300 K–345 K.

T_h	Thermoelectric cooler (CP2,31,06)								Heat sink (76620)		
	I = 4		I = 5		I = 6		I = 7		$Q_{hs} \text{ (conv)}$	$Q_{hs} \text{ (rad)}$	$Q_{hs} \text{ (total)}$
	Q_h	Q_l	Q_h	Q_l	Q_h	Q_l	Q_h	Q_l			
300.00	11.39	6.84	16.10	9.32	21.05	11.57	26.21	13.60	0.34	0.67	1.02
303.00	10.74	6.05	15.50	8.53	20.48	10.78	25.68	12.81	0.85	1.09	1.95
306.00	10.10	5.26	14.90	7.73	19.91	9.99	25.15	12.01	1.56	1.53	3.09
309.00	9.46	4.47	14.29	6.94	19.34	9.20	24.62	11.22	2.45	1.97	4.42
312.00	8.82	3.68	13.69	6.15	18.78	8.40	24.09	10.43	3.49	2.43	5.92
315.00	8.18	2.89	13.08	5.36	18.21	7.61	23.56	9.64	4.66	2.90	7.56
318.00	7.54	2.10	12.48	4.57	17.64	6.82	23.03	8.85	5.95	3.39	9.34
321.00	6.89	1.31	11.87	3.78	17.07	6.03	22.50	8.06	7.35	3.89	11.23
324.00	6.25	0.52	11.27	2.99	16.51	5.24	21.97	7.27	8.83	4.40	13.23
327.00	5.61	-0.27	10.66	2.20	15.94	4.45	21.44	6.48	10.40	4.93	15.33
330.00	4.97	-1.06	10.06	1.41	15.37	3.66	20.91	5.69	12.03	5.47	17.50
333.00	4.33	-1.85	9.46	0.62	14.80	2.87	20.38	4.90	13.73	6.03	19.76
336.00	3.69	-2.64	8.85	-0.17	14.24	2.08	19.85	4.11	15.49	6.60	22.09
339.00	3.04	-3.43	8.25	-0.96	13.67	1.29	19.32	3.32	17.30	7.19	24.49
342.00	2.40	-4.22	7.64	-1.75	13.10	0.50	18.79	2.53	19.15	7.80	26.94
345.00	1.76	-5.02	7.04	-2.54	12.54	-0.29	18.26	1.74	21.05	8.42	29.46

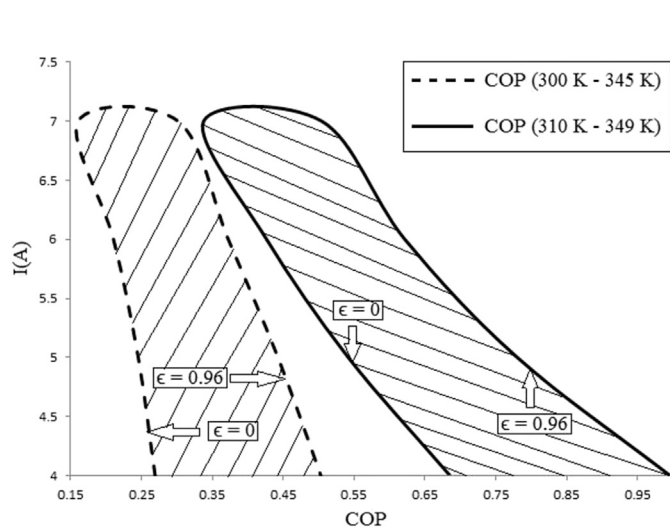


Fig. 10. Variation in COP with current values for both T_h ranges.

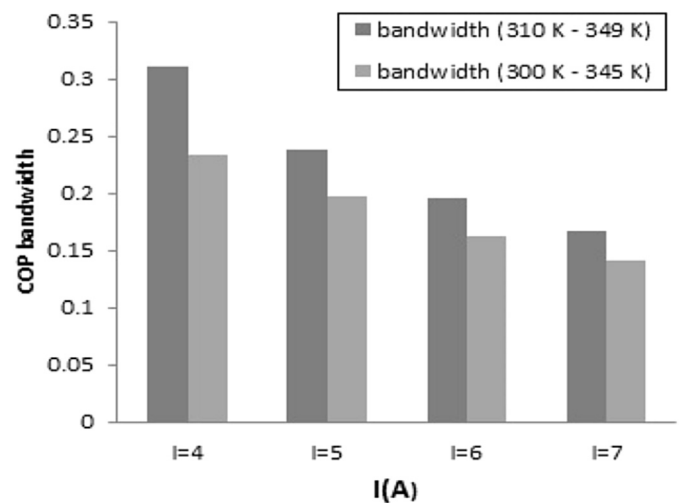


Fig. 11. Estimation of the difference in COP bandwidth between the two T_h ranges studied.

Table 4

Radiation contribution (in %) at various current levels.

I (current)	T _h (310 K–349 K)			T _h (300 K–345 K)		
	Q _h (conv)	Q _h (conv + rad)	% rad contribution	Q _h (conv)	Q _h (conv + rad)	% rad contribution
4.00	8.83	9.96	12.77	7.04	8.00	13.58
5.00	12.46	13.71	10.05	10.57	11.74	11.11
6.00	16.47	17.84	8.37	14.57	15.83	8.66
7.00	20.91	22.42	7.19	18.83	20.22	7.39

signifies that radiation effect which enhances the performance is comparatively less there.

Furthermore, the simulation is performed for the second range of T_h values (i.e. from 300 K to 345 K) with T_i set at 280 K keeping all the other parameters unchanged and varying the current as usual between 4 A–7 A alike the previous case. The similar plots are obtained to assess the role of heat sink thermal radiation onto the operating conditions of the TEC under the applied conditions and are shown in Fig. 7.

Fig. 7 shows the operating conditions of the TEC for the mentioned current range considering natural convection only (as $\epsilon = 0$) and radiation ($\epsilon = 0.96$).

The observations made over comparing similar Figs. 4 and 7 are presented through Fig. 8 to make the analysis more focussed.

It can be pointed out from Fig. 8 that there is a significant drift of the operating points (corresponding to temperature range i.e. 300 K–345 K) from the previous operating points (corresponding to temperature range i.e. 310 K–349 K) along both $\epsilon = 0$ and $\epsilon = 0.96$ curves. This is expected because the convective and radiative heat transfer decreases with the reduced working temperature values. The operating zone for radiation with the present T_h range (i.e. 300 K–345 K) is shown through Fig. 9. It is realized that most of the observations are similar to the previous case shown through Fig. 5 excepting that the non-functional zone under the present condition has come more closer to the operating region of radiation in compared to the previous case. Thereby it is clear that at lower T_h values, the TEC operations are safer at higher current ranges. This explanation has been further clarified, through data provided in Table 3. The COP of TEC has been computed using data from Table 3 and presented in Fig. 10.

The above figure demonstrates that the COP values are much higher for the first T_h range of 310 K–349 K in comparison with the second T_h range of 300 K–345 K, for both $\epsilon = 0$ and $\epsilon = 0.96$ at the same current level. This is mainly because in the latter case the T_i is set at a lower value (i.e. at 280 K) as compared to the ambient (i.e. at 295 K). Apart from this the COP bandwidth is also seemed to be more in case of T_h (310 K–349 K) than compared to the T_h (300 K–345 K), which is fundamentally because of the pronounced radiation effect in the high T_h range. This aspect is further made clear through bar diagrams given in Fig. 11.

It is quite clear that radiation has significant contribution to maximize the COP of the thermoelectric refrigeration process.

Table 4 reveals that for both the working temperature ranges, the percentage contribution through radiation decreases as the current value goes up. This further emphasizes that for higher current value, the gain in COP due to radiation will be less than compared to at lower current value.

4. Conclusion

The main motive of the present work is to bring modification into the earlier modelling of TEC by considering only natural convection as the sole mode of heat transfer from the heat sink. The consideration of the presence of surface radiation in addition to the natural convection as the mechanism of heat transfer, has been justified in this work. The COP for TEC improves significantly in presence of radiative heat fluxes. With the increase in the value of emissivity, the TEC can be operated at a much lower value of current. As the T_h value goes higher, the width of the radiation operating zone increases and also maintains a safe range from the non-functional zone and thereby ensuring smoother refrigeration. The variation of COP with respect to current, is much higher when the TEC works at comparatively larger values of T_h and T_i.

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